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Air conditioning system for a vehicle and associated operating method

The invention relates to an air conditioning system with a flow duct for an air stream to be conditioned and with a heat exchanger arranged in this flow duct and also with a circuit, operable in the heating or cooling mode, for the circulation of a fluid.

An air conditioning system of this type is used particularly in a motor vehicle. The refrigerant flow is in this case conventionally generated by a condenser or compressor which is inserted into the refrigerant circuit and is driven directly by the vehicle engine.

Modern low-consumption vehicles usually deliver too little waste heat or heating energy to make it possible to heat up the vehicle interior to comfortable temperatures in a time which, if required, may even be short. Particularly windshield defrosting lasts too long because of the low waste heat. In order to avoid this, it is known, for example from EP 0 960 756, to connect what may be referred to as a thermodynamic triangulation process in which a separate heat exchanger is provided for the additional heating of the air stream and therefore for conditioning. DE 3 907 201 also discloses an additional heat exchanger for heating the air stream. In addition, the air conditioning system known from EP 0 733 504 makes it possible to control and regulate the fluid or refrigerant circulating in the refrigerant circuit.

The air conditioning systems mentioned have the disadvantage that either they are not suitable for a circuit with carbon dioxide as fluid or refrigerant and consequently the heating capacity of such air conditioning systems is limited or that they require

additional components, in particular complicated and cost-intensive changeover or shutoff valves.

A carbon dioxide circuit, which is conventionally provided with a header or intermediate store arranged on the suction side and generally having the flow passing through it only in the cooling mode, is limited in its heating capacity, the latter moreover becoming lower with an increasing ambient temperature. This results from the dependence of the density of the vapor sucked in by the condenser on the ambient temperature. This leads to a reduction in the conveyed mass fluid flow or mass refrigerant flow and therefore also to a reduction in the heating capacity with a decreasing ambient temperature. Furthermore, in the heating mode, refrigerant and oil may accumulate in the intermediate store through which the flow does not pass, and because of this too little fluid or refrigerant flows through the circuit representing the heating mode. In order to avoid this, therefore, even in a carbon dioxide circuit, the circulating fluid stream is controlled according to requirements. This leads, in turn, to the use of complicated and cost-intensive regulating and control valves and also of additional lines.

The object on which the invention is based is, therefore, to specify a particularly simple air conditioning system for a motor vehicle, which allows as good a conditioning of an air stream as possible, along with a sufficiently good heating capacity. The object on which the invention is based is, furthermore, to specify a method for operating such an air conditioning system.

As regards a method for operating an air conditioning system, this object is achieved, according to the invention, by means of the features of patent claim 1.

As regards an air conditioning system, the object is achieved, according to the invention, by means of the features of claim 16.

The dependent claims relate to advantageous refinements and/or developments of the invention.

The main idea of the invention is to control a circuit having a fluid for conditioning an air stream in a heating mode in such a way that the intake pressure of a condenser at least partially overshoots a saturation pressure in the circuit caused by the ambient temperature, in the heating mode, the circuit preferably being operated in a dextrorotatory triangulation process, a drive power of the condenser being converted completely into heat by means of a heat exchanger, being transmitted to the air stream routed to the vehicle interior and thus being used for conditioning the air stream.

In a particularly advantageous development of the method, in the heating mode, the fluid in the circuit can be divided into at least one active part and at least one passive part.

Operating in a dextrorotatory triangulation process has the advantage that there is a high intake pressure and therefore a high mass flow in the circuit. In the method according to the invention, an intermediate store is incorporated into the heating mode, the fluid, for example a refrigerant, being supplied from the heat exchanger, for example a heating element, to the intermediate store, for example a low-pressure header, present in any case in the circuit, in order to flow through said intermediate store before being sucked in by the condenser.

By means of such a pressure-dependent control of the

fluid in individual regions of the circuit, in particular in a refrigerant circuit, the fluid quantity in the active part of the circuit is increased. In this case, depending on the type and design of the air conditioning system, particularly by virtue of appropriate methods of controlling and regulating the existing components, accumulated refrigerant can be recovered, as required, by being transferred into the as active part of the circuit. Such a control and regulation of the air conditioning system makes it possible to have a control and regulation of the heating capacity which is largely independent of the ambient temperature. Particularly by means of the accurate removal and introduction of refrigerant (= fluid) from the passive part and into the passive part of the circuit, the refrigerant stream circulating in the active part of the circuit can be set and optimized in terms of a predetermined heating capacity. Such a simple control and regulation of the refrigerant stream does not require any additional components, apart from the shutoff devices, control and/or regulating valves which are present in any case.

In one embodiment, the intake pressure can be controlled in a range of 10 bar to 110 bar.

In a further version of the method, with the activation of the heating mode, the fluid is routed out of the passive part of the circuit into the active part of the circuit. Additionally or alternatively, a threshold value for the intake pressure in the active part of the circuit may be predetermined, and, when said threshold value is undershot, the fluid is likewise routed out of the passive part of the circuit into the active part of the circuit.

To transfer the fluid out of the passive part of the circuit into the active part of the circuit, the

circuit operated in the heating mode is at least briefly changed over to the cooling mode or to a laevorotatory triangulation process. The changeover to the laevorotatory triangulation process has the advantage, as compared with the changeover to the cooling mode, that the laevorotatory triangulation process is likewise a heating process which is operated at a lower intake pressure than in the case of the dextrorotatory triangulation process.

The circuit is operated in the cooling mode or in the laevorotatory triangulation process up to the undershooting of a settable threshold value, the circuit being changed over to the heating mode again after the undershooting of the threshold value. The threshold value can be predetermined, for example, for an intake pressure and/or for a high pressure and/or for a hot-gas temperature at the condenser.

In an advantageous embodiment, the threshold value of the intake pressure is set at least 3 bar, preferably 5 bar, below the value of the saturation pressure caused by the ambient temperature.

Alternatively, the circuit can also be operated for a predeterminable period of time in the cooling mode or in the laevorotatory triangulation process, the circuit likewise being changed over to the heating mode again after the expiry of the period of time.

To increase the heating capacity, the air stream through the evaporator and/or through a gas cooler can additionally be reduced after the changeover to the cooling mode or to the laevorotatory triangulation process.

Since, in the case of commercially available electrically actuated 2/3-way valves, the magnetic

force is not sufficient to switch the valve when the differential pressure is too great, a pressure equalization is carried out in the circuit before the return to the heating mode.

In one embodiment, the circuit of the air conditioning system for a vehicle, in the heating mode, comprises a heat exchanger, an intermediate store and a condenser for the intermediate storage or for the condensation of a fluid, the condenser being operated at an intake pressure which is higher than the saturation pressure in the circuit caused by the ambient temperature.

In an advantageous refinement of the invention, an evaporator inserted in the flow duct of the air stream on the secondary side and in the circuit on the primary side is provided, in which case the evaporator can be connected in the circuit, on the exit side, to the intermediate store, with a nonreturn valve being interposed.

In an advantageous version of the air conditioning system, the volume of the evaporator for fluid reception is smaller than the storage volume of the intermediate store, the ratio of the storage volume of the intermediate store to the volume of the evaporator lying, for example, in the range of 2:1 to 20:1, preferably in the range of between 2:1 to 10:1.

To transfer the fluid out of the passive part into the active part of the circuit, and vice versa, the two parts of the circuit are connected to one another by means of at least one control device, the control device being opened in order to increase or reduce the fluid quantity in the active part of the circuit.

In a further advantageous embodiment, the condenser is connected to the evaporator on the exit side via a

control means and on the entry side via an associated controllable connecting line, after the opening of the control means gaseous fluid passing into the evaporator and forcing the liquid fluid located in the evaporator out of the evaporator into the active part of the circuit.

Exemplary embodiments of the invention are explained in more detail below with reference to a drawing in which:

figs. 1 to 3 show, in a diagrammatic illustration, alternative embodiments of an air conditioning system with a circuit, operable in the cooling or heating mode, for the return of a fluid flowing out from a condenser on the exit side and flowing into the condenser on the intake side via an intermediate store,

figs. 4 and 5 show thermodynamic graphs of the operation of the air conditioning systems according to figure 3.

Parts corresponding to one another are given the same reference symbols in the figures.

The air conditioning system 1 illustrated diagrammatically in figure 1 comprises a flow duct 4 through which an air stream 2 flows. In this case, the flow duct 4 has arranged in it an evaporator 6, in particular a refrigerant evaporator, which fills its cross section. In this case, for cooling the air stream 2 flowing into the flow duct 4 and flowing through the evaporator 6 on the secondary side, the evaporator 6 is connected to a circuit 8, forming a subcircuit 8A, for the circulation of the fluid F. The fluid F is, for example, carbon dioxide or another refrigerant. The circuit 8, by virtue of its functionality, is also designated as a refrigerant circuit. The subcircuit 8A is designated further as a passive subcircuit 8A

because of its passive routing of the fluid F for heating purposes.

The evaporator 6 is designed in the manner of a conventional refrigerant evaporator used in vehicle air conditioning systems (cf., for example, Kraftfahrtechnisches Taschenbuch/Bosch [Motor Drive Manual/Bosch] [Chief Editor H. Bauer], 23rd edition, Brunswick (Viebig), 1999, p. 777 ff.), in which heat is extracted from the air stream 2 flowing through owing to the evaporation of the refrigerant designated as the fluid F. To regulate the fluid F flowing through the evaporator 6, the evaporator 6 is preceded on the entry side by an expansion valve 12 which is arranged in the refrigerant circuit 8 and which can close sealingly.

The evaporator 6 is followed by a heating body 14, as seen in the flow direction of the air stream 2. The heating body 14 serves for the heating and therefore thermal control of the air stream 2 by means of coolant M heated by the engine 16. For this purpose, the heating body 14 is inserted on the secondary side into a coolant circuit 18. A coolant pump 20 for controlling the coolant stream is inserted in each case into the coolant circuit 18 on the entry side and exit side of the engine 16. In addition to the cooling of the coolant M, the latter is cooled by fresh air via a radiator 22 arranged in the air stream 102.

Furthermore, for the further heating of the air stream 2, the heating body 14 is followed in the flow duct 4 by a heat exchanger 24. The heat exchanger 24 is designed as a heating element and is inserted on the secondary side into a further subcircuit 8B of the circuit 8. The subcircuit 8B in this case causes an active control of the fluid F and is therefore designated further as an active subcircuit 8B. To control the fluid stream, an expansion valve 10 is

expediently inserted into the active subcircuit 8B between the condenser 26 and the heat exchanger 24.

In the cooling mode of the air conditioning system 1, a refrigerant runs through the passive subcircuit 8A according to the flow arrows P1 of the fluid F and therefore through the evaporator 6 and a condenser 26 driven by the engine 16. The fluid F is delivered in liquid form to the evaporator 6 and introduced. When it runs through the evaporator 6, the fluid F evaporates and at the same time extracts heat from the air stream 2 flowing through the evaporator 6 via corresponding heat exchange surfaces, not illustrated in any more detail. The fluid F, for example, a gaseous refrigerant, such as carbon dioxide, leaves the evaporator 6 and is supplied to a gas cooler 32 for cooling in the passive subcircuit 8A, with an intermediate store 28 and a heat exchanger 30 being interposed.

In the heating mode, the refrigerant runs through the active subcircuit 8B according to the flow arrows P2 of the fluid F, the fluid F being supplied on the exit side from the condenser 26 to the heat exchanger 24 designed as a heating element and being supplied to the condenser 26 again on the intake side via the intermediate store 28 designed as a low-pressure header and via the heat exchanger 30 which is deactivated in this mode. For changing over the flow of the fluid F from the active subcircuit 8B to the passive subcircuit 8A, or vice versa, shutoff devices 34 are arranged in the respective subcircuits 8B and 8A.

According to the present invention, the subcircuit 8B which is active in the heating mode makes it possible to convert the drive power of the condenser 26 into heat for the additional heating and thermal control of the air stream 2 by means of the heat exchanger 24, in

that the fluid F is supplied to the condenser 26 again on the intake side by the heat exchanger 24 via the intermediate store 28.

In order, as illustrated in figure 1, to avoid additional components for the air conditioning system 1, a suction pressure of the condenser 26 is in this case is set in such as way that the suction pressure at least partially overshoots a saturation pressure caused by the ambient temperature. The setting of the suction pressure is in this case brought about in a particularly simple way by means of structural features of the components of the air conditioning system 1. For example, for this purpose, a storage or evaporator volume representing the evaporator 6 is designed to be so small that the fluid quantity or refrigerant quantity collected or stored in the intermediate store 28 (= header) cannot condense completely in the cold evaporator 6, the shutoff device 12, for example an expansion valve, preventing a further outflow into the likewise cold gas cooler 32. Alternatively, the intermediate store 28 may have a correspondingly large storage volume which is substantially larger than the evaporator volume, the volume of the evaporator lying, for example, in the range of 50 to 500 ccm and the volume of the header lying in the range of 200 to 2000 ccm, so that a ratio of the volume of the header to the volume of the evaporator in the range of 2:1 to 20:1, preferably 2:1 to 10:1, can be selected.

Alternatively or additionally, as illustrated in figure 2, the air conditioning system 1 may be supplemented by a nonreturn valve 36 arranged in the passive subcircuit 8A on the exit side of the evaporator 6. The nonreturn valve 36 in this case prevents the fluid F or the refrigerant from being capable of flowing out of the active subcircuit 8B into the circuit components, namely the evaporator 6 and the

gas cooler 32, of the passive subcircuit 8A, said circuit components being substantially colder and therefore being under lower pressure in the heating mode. Since the condenser power is utilized for heating the air stream 2 purely due to the structural features of the components, an exact control of the fluid stream in the active subcircuit 8B is not possible.

However, for example, after the vehicle has been stopped several times or stopped for too long, an exact control and setting of the circulation of the fluid F in the heating mode is desirable, if not absolutely necessary, since, in this case, too much fluid F collects in components, namely the evaporator 6 or the gas cooler 32, of the passive subcircuit 8A and therefore the performance of the circuit 8 is markedly limited. In the other situation where too much fluid F flows in the active subcircuit 8B, the suction pressure may, under certain circumstances, rise to too high a value, with the result that the condenser 26 may be damaged.

For this purpose, as illustrated in figure 3, two control devices 38A and 38B are arranged in the active subcircuit 8B. The control devices 38A and 38B are designed, for example, as regulating valves or expansion valves. Depending on the type and activation of the control devices 38A and 38B, various operating processes of the air conditioning system 1 in the heating mode can be set, which are explained in more detail by means of the thermodynamic graphs according to figures 4 and 5.

Figure 4 in this case shows a pressure/enthalpy graph for what is known as a laevorotatory triangulation process, and figure 5 shows a pressure/enthalpy graph for what is known as a dextrorotatory triangulation process. According to the method for operating the air

conditioning system 1, as shown in figure 4, the fluid F flowing out of the condenser 26 at high pressure is supplied, throttled only insignificantly, directly to the heat exchanger 24 by means of the control device 38A according to the curve K1, according to curve K2 the fluid F discharging its heat to the air stream 2 flowing through the heat exchanger 24 on the primary side. The fluid F is supplied to the intermediate store 28 and the condenser 26 again from the heat exchanger 24 via the control device 38B which, according to the curve K3, throttles the fluid F to the intake pressure. For this purpose, the control device 38A, designed as an expansion valve, is opened as completely as possible, so that the pressure loss is low, and the predominant pressure reduction is executed at the control device 38B, likewise designed as an expansion valve. The curve K4 illustrates the pressure increase of the fluid stream brought about by means of the condenser 26.

By a change in the width or degrees of opening of the control devices 38A and 38B, the laevorotatory triangulation process according to figure 4 can be changed over to a dextrorotatory triangulation process according to figure 5. For example, a changeover from the dextrorotatory triangulation process to the laevorotatory triangulation process takes place in the situation where too little fluid F or refrigerant, for example what is known as R744 refrigerant, flows in the active subcircuit 8B. In the dextrorotatory triangulation process, the substantial pressure reduction takes place at the control device 38A according to figure 5, curve K1, while the control device 38B is completely open and therefore brings about only a low pressure reduction to the value of the intake pressure, according to curve K3. The result of this, in a comparison of the laevorotatory with the dextrorotatory triangulation process, is that the

values of the intake pressures in the dextrorotatory triangulation process according to figure 5 are higher than in the laevorotatory triangulation process according to figure 4. On account of this, the condenser 26 can convey a greater mass flow of the fluid F, with the result that additional heating capacity is generated by means of the heat exchanger 24. If the switch to the laevorotatory triangulation process is made, the intake pressure falls considerably, as compared with the dextrorotatory triangulation process. If this intake pressure in the laevorotatory triangulation process falls below the pressure in the passive system part caused by the ambient temperature, a displacement of refrigerant from the passive part into the active part occurs.

In the event that too much fluid F flows through the active subcircuit 8B, a further control device 38C between the two subcircuits 8A and 8B is switched. By the control device 38C being opened, the fluid F can then be routed, metered correspondingly, into the passive subcircuit 8A to the gas cooler 32 or to the evaporator 6.

In a further instance of use, to eliminate an accumulation of fluid F or refrigerant, for example after a lengthy standstill, at the start of travel the vehicle is operated, during the run-up of the condenser 26, in the cooling mode and therefore in the passive subcircuit 8A within a predetermined time range, with the shutoff device 12 open at maximum, in order to bring about a sufficiently good throughflow of the gas cooler 32, of the evaporator 6 and of the intermediate store 28. As a result, an accumulation of fluid F in the evaporator 6 or else in the gas cooler 32 is at least partially eliminated, in that a sufficient quantity of the fluid F, in particular of liquid refrigerant, is routed into the intermediate store 28

or header and stored. Subsequently, the air conditioning system 1 is operated according to one of the above-described triangulation processes, as shown in figure 4 or 5. If, in this case, an accumulation of fluid F in the passive subcircuit 8A occurs again, the heating mode (also called additional heating operation) of the active subcircuit 8B is interrupted by means of a brief changeover to the cooling mode of the passive subcircuit 8A which routes the fluid F into the intermediate store 28 again.

In addition to the control of the air conditioning system 1 according to the triangulation processes, as shown in figures 4 and 5, by means of the above-explained design of the components and/or arrangement of the control devices 38A to 38C, the condenser 26 may be provided on the intake side with a pressure sensor, not illustrated in any more detail. The pressure sensor in this case serves for the essentially exact determination of the fluid quantity in the active subcircuit 8B, thus making possible an accurate and therefore settable introduction or removal of the fluid F between the two subcircuits 8A and 8B. In another alternative, the condenser 26 is connected on the exit side to the evaporator 6 on the entry side via a controllable connecting line 40.

List of reference symbols

1	Air conditioning system
2	Air stream
4	Flow duct
6	Evaporator
8	Refrigerant circuit
8A	Passive subcircuit
8B	Active subcircuit
10	Expansion valve
12	Sealingly closing expansion valve
14	Heating body
16	Engine
18	Refrigerant circuit
20	Coolant pump
22	Radiator
24	Heat exchanger
26	Condenser
28	Intermediate store
30	Heat exchanger
32	Gas cooler
34	Shutoff devices
36	Nonreturn valve
38A,38B,38C	Control device
40	Connecting line
42	Control device
102	Air stream
F	Fluid
M	Coolant
K2,K3	Curve
P1,P1	Flow arrow